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INTRINSICALLY IRREVERSIBLE HEAT ENGINES

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I would like to begin by saying that the remarkably simple and beautiful experiments on liquid ^3He and on ^3He-^4He solutions done at Duke University by Bill Fairbank, King Walters, and others in the decade of the fifties helped lay the foundations for a field which is very active today. King Walters will describe those days better than I. But I want to add how important those early observations of Fermi-like behavior in liquid ^3He and of the phase-separation phenomenon in ^3He-^4He solutions have turned out to be and how they provided something secure on which I could build my own experimental work in the field of liquid helium.

My contribution below to this conference differs from most of the others, yet I feel that it is still appropriate. Certainly one of the themes of the Near Zero Conference is the importance to physics of attaining temperatures near the absolute zero. Although there is no longer any problem in reaching exceedingly low temperatures, in some sense the motivation for my present far-from-complete work is to reexamine old ideas and investigate new ones to see if it might not be possible to attain low temperatures <u>more simply</u>, and in doing so to open up the temperature region "near zero" to even more widespread discovery and invention. In this work I am only following Carnot's admonition, written in the last paragraph of his book,[1] to the effect that there is more to the subject of heat engines[2] than the efficient use of heat and work; that convenience, economy, and above all simplicity are essential components of the solution to any practical problem. I like to think that some of the hallmarks of Bill Fairbank's contributions to physics are both the ingenuity and the essential simplicity of his approach to the solution of practical measurement problems.

I. INTRODUCTION

Only heat engines which work on a long time scale can have the Carnot efficiency. If power, or work (or heat) per unit time, is the important quantity then irreversibilities are unavoidable. For example, a Carnot-like engine with a finite thermal resistance between working fluid and reservoirs has the diminished efficiency at optimal power output[3,4] of $1-(T_C/T_H)^{1/2}$

rather than $1-(T_C/T_H)$, where T_C and T_H are the temperatures of the cold and hot reservoirs.

An example of an engine which has Carnot's efficiency in the limit of long periods is the Stirling engine.[5] Rev. Robert Stirling invented this reciprocating engine in 1816, eight years before Carnot's book was published. I think of Stirling's engine as being intrinsically reversible. Stirling introduced into the engine what I call a second thermodynamic medium in addition to the primary thermodynamic medium (air for Stirling's engine). Engineers call this second medium a heat regenerator. It plays the role of a continuously distributed heat source and sink internal to the engine and makes possible an idealized cycle having Carnot efficiency in which all processes are locally isothermal. The Stirling engine, as for all intrinsically reversible engines, has two separately controlled elements; in this case a power piston and a displacer which are phased appropriately with respect to one another to produce a useful output.

The intrinsically irreversible heat engines I discuss here use the irreversible process of thermal conduction to achieve the necessary phasing between temperature changes and motion of a primary medium and therefore have only one moving mechanical element. Stirling's second thermodynamic medium is also an essential element of such engines, but now the thermal contact with it is neither very good nor very bad. Indeed, no useful output is obtained either in the reversible isothermal limit, where the engine runs very slowly; or in the reversible adiabatic limit, where the engine is running very fast. That is why I call such engines "intrinsically irreversible." It is also necessary for the geometrical symmetry between primary and secondary media to be broken in order to achieve a useful thermodynamic result.

If an external source works on such an engine then a temperature difference is produced. If the primary medium is a thermodynamically active fluid with a very low Prandtl number, so that effects of viscosity are negligible, then I think that very large temperature differences should ensue. As it is possible to design engines in which thermodynamic effects are emphasized over frictional effects, I am encouraged to pursue a scientific study of intrinsically irreversible engines with the hope that there will be practical consequences, especially for the simple attainment of low temperatures. Essential to the simplicity are the need for only one moving element and the quality that thermal contact to the second medium must not be good.

I refer to the present class of engines as intrinsically irreversible but functionally reversible to distinguish them, for example, from refrigerators which employ the irreversible Joule-Thomson expansion of a fluid to produce low temperatures. The Joule-Thomson, or perhaps better the Linde-Hampson, cooling method is characterized by simplicity and effectiveness and indeed was the means employed by Kamerlingh Onnes to liquify helium. The present engines are functionally reversible in the sense that they can be prime movers as well as refrigerators, just depending on the temperatures spanning the engine. As prime movers, in one form they are the famous Sondhauss tubes in acoustics[6]. In another form familiar to all low temperature experimentalists they lead to the "Taconis oscillations" in a tube inserted through a temperature gradient into a vessel containing liquid helium.

The possibility of using the irreversible process of thermal conduction as part of a method to produce cold was first proposed by Gifford and Longsworth and then developed by them in a series of papers.[8-11] They called their device a "pulse tube refrigerator" and described its operation in terms of a concept called "surface heat pumping." It operated at low frequency (ca. 1 Hz) and used pressure changes of several atmospheres. I was reminded of the Gifford and Longsworth papers last year when I ran across them again while browsing in the library at the Kamerlingh Onnes laboratory. Just at that time I was also studying a paper by Ceperley[12] on a traveling wave acoustical engine. The Ceperley engine is essentially a novel Stirling engine. It requires, as for all Stirling engines, regions of high thermal contact between primary and secondary media, and it suffers badly from effects of viscous dissipation. However, combining acoustical techniques with thermodynamically irreversible processes seemed like an eminently sensible thing to do. So on my return to Los Alamos a year ago, and with the help of my colleagues, Greg Swift and Al Migliori, I began studying intrinsically irreversible acoustical heat engines experimentally. The practical qualities of simplicity, high frequency (for a heat engine), automatic provision of a "flywheel" by the inertia of the gas, absence of moving seals, and dependence on poor thermal contact were immediately obvious. Only more recently have I appreciated the generality of the concepts.

II. THE PRINCIPAL THERMODYNAMIC EFFECTS

An apparatus in which the principal thermodynamic effects can be observed is shown in Fig. 1. I want to emphasize that from a fundamental point of view the use of acoustical techniques in this apparatus is peripheral. An acoustical compression driver (loud speaker) excites a resonant acoustical tube in the form of a U which is closed at one end. The U shape is used to eliminate natural convective heat transfer. The second thermodynamic medium, on which our attention will be focused, is a stack of 19 fiberglass plates shown in more detail in Fig. 2. These are instrumented by epoxying five chromel-constantan thermocouples TC1 to TC5 to the central plate with the junctions located more or less as shown. The tube contains ^{4}He gas at an average pressure p that we can control. At the closed end of the tube the dynamic pressure amplitude P_0 can be measured accurately. Both the driver and the closed end of the tube, whose temperature is measured by thermocouple TC6, are near room temperature. The walls of the straight sections of the U-tube are 1 mm thick fiberglass and are insulated with superinsulation, the whole U-tube being in vacuum. The bottom of the U-tube is copper whose temperature, measured using TC7, can be adjusted by controlling the flow of N_2 boiloff gas from a liquid N_2 bath. The open space above and below the second medium is designed to have a high thermal resistance. This form of the apparatus, with gas cooling at the bottom of the U, is necessary to reduce the heat input to the bottom end of the second medium to a controllably small value. The tube is operated at resonance. Although heat generated in the second medium itself by viscous effects is small, owing to its proximity to the pressure antinode, heat generated viscously in other parts of the tube and heat from the room temperature region is transported toward the second medium, in part by acoustical streaming. In a practical cooling engine it will be necessary to play a trick to eliminate this problem, but for the present purposes of illustrating the principles the arrangement shown is satisfactory. As the length of the second medium is short compared to the radian length of the acoustical wave, whose frequency is typically ca. 400 Hz, the same results would presumably be achieved ideally in another shorter apparatus with an oscillating piston located close to the second medium. Acoustical techniques are used here primarily for practical reasons.

In this apparatus we can measure temperature and static and dynamic pressure. Two types of experiments give readily interpretable results. In one, with the apparatus initially at a uniform temperature (usually ambient), the acoustical power is suddenly applied for a short time and the thermocouple response measured. In such a measurement the initial time rate of change of temperature T_1 is proportional to the heat flux into the surface of the second medium. In a second experiment the temperature distribution along the second medium is observed for essentially adiabatic conditions under the action of continuous wave acoustical power.

Typical results of the sudden application of acoustical power are shown in Fig. 3. They show the essence of the thermodynamic effects. Before acoustical power is applied, we have $(\text{grad}_z T) = 0$ along the plates. The ^4He pressure is 4.90 bar and the sound frequency is adjusted to resonance, near 400 Hz. At time $t = 0$ acoustical power is applied with amplitude ratio $P_0/p = .0092$. This corresponds to only about 4% of a typical power level and was adjusted for demonstration purposes to allow all five thermocouple responses to be shown with the same sensitivity. Each of thermocouples 2, 3, and 4, those within the plates, had very nearly the same response, a heating with T constant but not large. Thermocouple 5, located at the edge of the plates closest to the closed end, heats rapidly while 1, located at the edge of the plates closest to the driver end, cools even more rapidly. These observations are consistent with the following time-averaged energy flows in the gas toward the closed end. Within the plates but far from the ends, the energy flow toward the closed end increases linearly with the distance z from the closed end. Then, from conservation of energy, the time-averaged heat flow per unit length into the plates will be constant, as observed. There are sudden changes in the energy flow at the ends of the plates, the energy flow in the gas being smaller at both ends outside the plates. Then, at TC1, the energy flow toward the closed end is greater within than it is outside the plates, so heat must be extracted from the plates and their temperature must drop, as observed. At TC5 the energy flow toward the closed end is greater within than it is outside the plates, so the plates must absorb heat, their temperature rising as observed. The thermodynamic action corresponding to the heating at TC5 and the cooling at TC1 is a central feature of the irreversible engine.

The second experiment which can be done reasonably well is to study the adiabatic temperature distribution. Here what is done is to let the acoustical power act continuously, adjusting the temperature at TC7 and observing the temperatures of TC1 through TC5. As I pointed out in describing the apparatus the thermal resistance of the spaces adjacent to the stack of plates is high, so the response of the plates is nearly adiabatic. Typical results of this experiment for ^{4}He gas at p = 1.90 bar and a dynamic pressure ratio P_0/p = 0.04 are shown in Fig. 4, where the absolute temperatures of the five thermocouples are shown vs. their longitudinal positions on the stack. The data are obtained as follows. Starting with the whole stack at the ambient temperature, that of TC6 at about 287° K, the acoustical power is turned on. There is an immediate response: in a few minutes a quasi-equilibrium temperature distribution is developed with $T_5 > T_6$ and $T_1 < T_6$ and a temperature difference (T_5-T_1) comparable to that on any of the curves shown. Then depending on how T_7 is manipulated we find a variety of very similar equilibrium distributions, four of which are shown in Fig. 4. A good way to describe this distribution is "rigid." It is established quickly. The general temperature level can be changed slowly by changing T_7, but this control is very "soft." The distribution is insensitive, at a level of 5 to 10%, to variations of static pressure over a factor of 10 (from 0.5 to 5 bar), to variations of dynamic pressure P_0/p over a factor of 5 (and thus in power over a factor 25), and to small variations in frequency (factor 2 to 3). It also does not depend strongly on either T_6 or T_7. The dashed line on Fig. 4 is calculated using the adiabatic equation of state for ^{4}He gas, with T being the temperature at z, with V being the volume included between the closed end of the tube and the point z, and with a forced fit at TC5. A relationship like this between the adiabatic equation of state of the gas and the temperature distribution in absence of heat flows was proposed by Gifford and Longsworth.[9]

III. UNDERSTANDING THE INTRINSICALLY IRREVERSIBLE ENGINE WITH IDEAL GAS PRIMARY WORKING SUBSTANCE.

The concept of phase is important in heat engines. As shown in Fig. 5a, the phasing in an internal combustion engine is provided by the correct timing of the ignition or injection of fuel into the cylinder with respect to the motion of the piston. The resulting indicator diagram (or p-V plot) has an enclosed area corresponding to the work done in a cycle. Proper phasing is achieved in a Stirling engine, Fig. 5b, by moving the piston P and the displacer D independently to maximize the area of the indicator diagram. In each of the above two quantities had to be controlled and caused to act in proper time sequence to achieve the desired result. In Fig. 5c I consider a piston and cylinder enclosing a gas with only one moving element, the piston, and with only reversible processes considered. If τ is the thermal relaxation time of gas in the cylinder and ω is the angular frequency of piston motion then the two possibilities for reversible processes are either isothermal ($\omega\tau << 1$) or adiabatic ($\omega\tau >> 1$). Starting at the same state (p,V) and letting the piston oscillate produces no included area on the indicator diagram in either extreme. But if, as suggested in Fig. 5d, the thermal contact is too poor to allow isothermal processes yet not so poor as to make the processes adiabatic, then the indicator diagram will have a net included area. In this case, and for a small sinusoidal variation of volume with $\omega\tau < 1$ so that the processes are not quite isothermal, the indicator diagram will appear as shown. During expansion the gas will be generally lower in temperature than its surroundings, and hence at a lower pressure than the isotherm; during compression the gas will be generally higher in temperature and hence at higher pressures than the isotherm. Hence with only one moving element, the piston, and irreversibility there is a finite included area on the indicator diagram. But that is not enough to produce an interesting thermodynamic effect.

Excepting the region very near to the piston itself the above irreversibility just leads to the absorption of heat by the walls of the cylinder. What is necessary to produce something interesting is to make the symmetry of the second medium and of the space occupied by the primary medium different. Why this is important can be understood rather easily by considering energy flow in the gas. Refer to Fig. 6a, which shows a cylinder with walls at constant temperature containing a gas confined by an oscillating piston. Excluding kinetic energy of mass motion the energy flow in the gas is the enthalpy flow. What is of interest is the time-averaged energy flow $\bar{\dot{H}}$, taken positive in the direction along z toward the closed end. If $\bar{\dot{H}}$ depends on x then $-d\bar{\dot{H}}/dx$ is, by conservation of energy, the time-averaged heat flow per unit length into the walls of the cylinder. The time-averaged energy flow can be written schematically

$$\bar{\dot{H}} = \overline{\overline{A v \rho h}} = \overline{\overline{A\ v\ \rho c_p \delta T}} \ , \qquad (1)$$

where the double bar indicates both a time and spatial average, A is the cross-sectional area, ρ is mass density, c_p and h are specific heat and enthalpy per unit mass and δT is the temperature deviation from the isothermal surroundings. For isothermal processes, the temperature deviation δT is zero; while for adiabatic processes v and δT have a $\pi/2$ phase difference, so that their time-averaged product is zero. Hence for both types of reversible processes $\bar{\dot{H}}$ is zero. In the limit of zero Prandtl number and where the thermal penetration depth

$$\delta_\kappa = (2\kappa/\omega)^{1/2} \qquad (2)$$

is small compared to the distance between solid walls, where $\kappa = K/\rho c_p$ is the thermal diffusivity, K is thermal conductivity, and ω is angular frequency of the piston motion; the energy flow can be calculated simply with the result that $\dot{H} \propto v_0 P_0 \Pi \delta_\kappa$, with proportionality constant of order unity. From (1) the area A is replaced by the product of the perimeter (at z) Π and the penetration depth δ_κ, the velocity v by the velocity amplitude (at z) v_0, and the quantity $\rho c_p \delta T$ by P_0, the amplitude of the dynamic pressure variation. [For ideal gases $(\partial T/\partial p)_s = (\rho c_p)^{-1}$.] Expressing v_0 in terms of the compressibility, the distance $|z|$ from the closed end, the angular frequency ω, and P_0; and putting in the numerical constants valid for ^{4}He gas one gets

$$\bar{\dot{H}} = \frac{1}{4}\ \omega \delta_\kappa p \left(\frac{P_0}{p}\right)^2 \Pi\ z \quad . \tag{3}$$

The energy flow depends on both perimeter, or surface area per unit length, and distance from the closed end. For a geometry such as that of Fig. 6b, where a set of plates has been inserted into the cylinder, the dependence of energy flow on $|z|$ is as shown: moving away from the closed end there is a sudden increase where the plates begin, a steady increase over the region of the plates, and then a sudden decrease where the plates end. $\bar{\dot{H}}$ does not drop to zero outside the plates as of course work is being performed on the gas in the plates and beyond. (There are no <u>sudden</u> changes in $\bar{\dot{H}}$, but rather rapid changes over distances estimated by the amplitude of the fluid motion.) These are exactly the qualities expected from the experimental data shown in Fig. 3. Absent the symmetry breaking geometrical change, only heat rejection to the second medium would result.

If the above is somewhat unfamiliar, then consider for purposes of understanding an analogous magnetic system. Imagine a magnetic body with a system of spins (the primary medium) and a lattice (the second medium) coupled by a thermal resistance. Let the magnetic body move in a sinusoidal fashion at angular frequency ω in an inhomogeneous field so that temperature changes will result. If τ is the spin-lattice relaxation time, then there is no <u>average</u> heat consequence either in the isothermal ($\omega\tau \ll 1$) or adiabatic ($\omega\tau \gg 1$) limits. But in between, for $\omega\tau \sim 1$, we know that the magnetization and field will be phased so that on the average heat is rejected to the lattice. There is, however, no useful thermodynamic result; only the direct conversion of work into heat at the lattice temperature. This corresponds to the situation in the cylinder of Fig. 6a before the plates are introduced. But now suppose that the "spins" and the "lattice" are in <u>different</u> bodies coupled by some thermal resistance and let the symmetry between them be broken by, say, limiting the size of the "spins." Furthermore let the "spins" be moved in an oscillatory fashion with respect to the "lattice" in the inhomogeneous magnetic field. Then what will happen is that the end of the "spins" in the weaker field will cool while that end in the stronger field will heat. A useful thermodynamical effect has been produced which will be optimal in the vicinity of $\omega\tau \sim 1$.

The above considerations were based on a starting condition of uniform temperature with $(\mathrm{grad}_z T) = 0$. Suppose we let the engine run continuously so that a steady temperature difference is achieved. In a practical engine this temperature difference will be limited by for example external heat flows, internal heat generation due to viscosity, longitudinal conductivity of the various thermodynamic media and of the container, and heat transport due to natural or forced convective motions of the fluid. The ideal fluid has no viscosity but does have thermal conductivity; it has zero Prandtl number. In a typical fluid the effect of viscosity need not be overwhelming as the refrigeration effect at the end of the plates is proportional to the plate length while the overall viscous heating effect is proportional roughly to the cube of the plate length; I think that the two can be adjusted appropriately by choice of geometry and frequency so that viscous effects will not be large. For the ideal fluid in the ideal engine in the absence of external heat flows the temperature gradient will continue to develop as acoustical power is applied until heat flow to the second medium drops to zero. To understand this consider a particle of fluid shown in a cylinder in Fig. 7. As it moves toward the closed end its temperature rises and as it moves away its temperature falls. Initially, $(\mathrm{grad}_z T)$ is zero, and a given fluid particle tends to give heat to the walls on compression and take heat from the walls on expansion. But as the engine runs and T increases toward the closed end the temperature difference driving the heat flow to the second medium decreases. Finally, at some limiting temperature gradient in the wall $(\mathrm{grad}_z T)_{\lim}$ the heat flow to the wall will reach zero and we will have the condition

$$x_0 \left|(\mathrm{grad}_z T)\right|_{\lim} = \delta T_0 \quad , \qquad (4)$$

where x_0 is the amplitude of the particle motion and δT_0 is the amplitude of the adiabatic temperature change of the fluid particle. For adiabatic processes in an ideal monatomic gas $\delta T_0/T = -(2/3)\,\delta V/V$ and $P_0/p = -(5/3)\delta V/V$. As the pressure and pressure changes are uniform even in the gas with non-uniform temperature, the fractional volume change amplitude $\delta V/V$ of a given fluid particle at z is the same as the total fractional volume change amplitude $A(z)\,x_0/V(z)$ beyond z, where $A(z)$ is the cross-sectional area at z and $V(z)$ is the volume included between z and the closed end. As a consequence, from (4)

$$\left|\mathrm{grad}_z T\right|_{\lim} = \frac{2}{3}\,\frac{T}{V(z)/A(z)} \quad . \tag{5}$$

This is equivalent to

$$T(z)\;V(z)^{2/3} = \text{constant} \tag{6}$$

independent of pressure amplitude, pressure, and frequency. This result, which was derived by Gifford and Longsworth[9] in a similar way, gives a reasonable accounting of the results on Fig. 4. The "rigid" temperature distribution found there is essentially independent of the various experimental parameters and seems to be correlated with the adiabatic equation of state of the gas.

Heat flows into plates at $(\mathrm{grad}_z T) = 0$ have been studied extensively by us and compared with a thermoacoustical theory by Rott,[14] with excellent agreement where comparision is appropriate. The behavior of the engine in the "adiabatic limit" is reasonable. We have not yet dealt quantitatively with the qualities of the engine in the presence of both substantial heat flows and a temperature gradient, nor have we attempted prime-mover operation. Hence we are not yet able to describe the engine's efficiency under prescribed conditions. Since for a prime mover heat must be absorbed at the hot end and rejected at the cold end, the prime mover and refrigerator functions of this engine are separated by the temperature distribution corresponding to the adiabatic limit. Only if the temperature gradient exceeds that given by

Eq. (5) will the gas on compression be cooler than its immediate surroundings so that heat can be absorbed. For refrigerator operation heat transfer to the hot reservoir and from the cold reservoir can be via regions of constant temperature. But for prime mover operation some of the overall temperature difference available must be used up to achieve suitably large temperature gradients to achieve the necessary direction of heat transfer. This could be looked upon as an undesirable quality, but then it could also mean greater simplicity of the necessary heat exchangers. And simplicity after all is one of the most important practical characteristics of these engines.

IV. GENERAL REMARKS

The engine as we have been describing it really does not depend on acoustical concepts. But as simplicity is such an important quality of intrinsically irreversible engines, and as acoustical techniques are so simple and elegant and the relatively high frequency useful, future study of an acoustical engine is very desirable. But there are problems to be overcome. One is concerned with viscous heating. To reduce viscous heating to a small value relative to the thermodynamic cooling effect it is essential that the second medium be placed near a pressure antinode and that the ratio of the length of the second medium to the radian length of the acoustical wave be suitably small. Even then the heating effect of the remainder of the open resonant tube is excessive. A way must be found to deal with this problem. A second problem is associated with the effects of acoustical streaming.[3] In this well known effect an oscillatory flow in the presence of boundaries leads to a steady circulational or forced convective flow. As a consequence we can expect that $|(\text{grad}_z T)|$ will be decreased. I suspect that this is the reason why we have not been able to achieve the full adiabatic temperature difference in experiments like that which produced Fig. 4. No doubt even more interesting problems will appear as we go more deeply into this subject.

The general configuration of the second medium that we have used, as in Fig. 6b, could be changed by letting it extend fully to the closed end of the tube. In that case one might want to use the last bit to make contact with the hot reservoir. But suppose that we chose instead to lengthen the open space at the end of the tube. Now in the way we have operated the engine, if τ_p is the thermal time constant in the region of the plates and τ_o is the thermal time constant in the open region, then we normally have $\omega\tau_o >> 1$ and $\omega\tau_p \approx 1$. Processes in the open space are essentially adiabatic while in the region of the plates thermal contact is just poor. The energy flow curve of Fig. 6b applies. However, if for the same geometry we decrease ω until the thermal penetration depth in the open tube is comparable to the radius, then we will have $\omega\tau_o \sim 1$ and $\omega\tau_p << 1$. The time-averaged energy flow in the open space goes up; that in the plates decreases toward zero. The <u>signs</u> of the discontinuities in $\bar{H}$ at the ends of the plates change; the regions of heating and cooling are interchanged. It is interesting that this latter condition is the one that probably applied to the experiments of Gifford and Longsworth. In any case more than geometrical configuration is important in these engines.

I have said before that I think the concepts presented here are quite general, not limited to any particular working substance. Whether or not they are used in any practical case probably will be related to the importance of simplicity. The essential requirements seem to be (1) both primary and secondary thermodynamic media, (2) relative motion between primary and secondary media, (3) an irreversible process, preferably thermal, to provide suitable phasing between motion and temperature changes, and (4) a breaking, either continuously or discontinuously, of the relative symmetry between primary and secondary media. The primary medium must be thermodynamically active; that is, its entropy at constant temperature should depend adequately strongly on some externally applied parameter such as pressure or magnetic field. The secondary medium should have an adequate heat capacity.

The general principle which determines the steady state operation of this type of engine has not been stated. It probably should be consistent with the principle of minimum rate of production of entropy,[15] at least for the present experiments. Thus, for the ideal engine with no external heat input and where the thermodynamic effects are due to thermal conduction and viscous effects are negligible, as the engine operates and the temperature gradient develops the rate of production of entropy decreases. Finally, in the dynamic equilibrium state when heat transfers to the second medium cease the rate of production of entropy has been reduced to zero.

The intrinsically irreversible heat engine is certainly a conceptually and scientifically interesting object. Whether or not it will lead to the hoped-for solutions to practical problems in low temperature technology has yet to be shown. The limiting temperature distribution of Eq. (6) should apply provided both that sources of heat can be eliminated and that $\left|(\mathrm{grad}_z T)\right|$ is not so large as to lead to an excessive heat transfer within the second medium. From Eq. (6) factors of 2, 4, and 8 reductions in temperature require, respectively, factors of 2.8, 8, and 27.6 in volume ratio. It is also possible in principle to provide stages of temperature reduction as is common in cooling using more conventional cooling methods. Or one might combine an irreversible acoustical engine with a Joule-Thomson engine or with, say, an intrinsically irreversible magnetic engine. For the present a detailed scientific understanding of intrinsically irreversible heat engines and the problems and pecularities of their acoustical manifestations must precede practical developments.

V. ACKNOWLEDGEMENTS

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FIGURE CAPTIONS

Fig. 1. Schematic diagram of apparatus using acoustical techniques to study the qualities of an intrinsically irreversible heat engine and using ^{4}He gas near room temperature as primary medium and fiberglass plates as secondary medium. TC1 through TC7 are thermocouple sensors, p is the static pressure and P_0 the amplitude of the dynamic pressure.

Fig. 2. Schematic drawing showing the geometry of the parallel plate second medium and of the thermocouple locations.

Fig. 3. Essential thermodynamic effects in the engine. Shown are superimposed chart recordings of thermocouple temperature changes vs time. There is no initial longitudinal temperature gradient: $(\mathrm{grad}_z T) = 0$. At $t = 0$ the acoustical power is suddenly switched on. For $t > 0$ the acoustical power is constant. The numbers refer to the thermocouple locations given on Figs. 1 and 2.

Fig. 4. Absolute temperature distributions as obtained from dynamic equilibrium values of the five thermocouple outputs for $p = 1.90$ bar and $P_0/p = 0.04$. These four curves were obtained for four different values of T_z, Fig. 1, the lowest curve being for the lowest value of T_z. The point $z = 0$ corresponds to the end of the plates where TC5 is located. The line labeled $TV^{2/3}$ = constant is based on theoretical arguments presented in Sec. III, where T is the temperature at z and V is the gas volume included between z and the closed end of the tube.

Fig. 5. Illustration of how phase plays a role in heat engines. p and V refer to the pressure and volume of gas in the cylinder. (a) refers to an internal combustion (I.C.) engine with heat introduction at the (*). (b) shows a Stirling engine with piston P and displacer D moved individually in an articulated cycle. (c) describes reversible isothermal ($\omega\tau \ll 1$) and reversible adiabatic ($\omega\tau \gg 1$) processes; ω is angular frequency and τ is thermal relaxation time. (d) shows the behavior for an intermediate thermal contact with thermal penetration depth δ_κ comparable to the cylinder radius.

Fig. 6. (a) Geometry used for the calculation of energy flow in the gas. z and v are local gas position and velocity. δ_κ, Eq. (2) in text, is the thermal penetration depth. (b) Time-averaged energy flows as a function of z for $(\mathrm{grad}_z T) = 0$ when a stack of plates is introduced into the cylinder of (a).

Fig. 7. A particle of gas oscillates simple harmonically in the cylinder at distance z from the closed end with amplitude x_0 and temperature variation of amplitude δT_0. The induced longitudinal temperature gradient in the second medium is $(\mathrm{grad}_z T)$.

SCHEMATIC APPARATUS

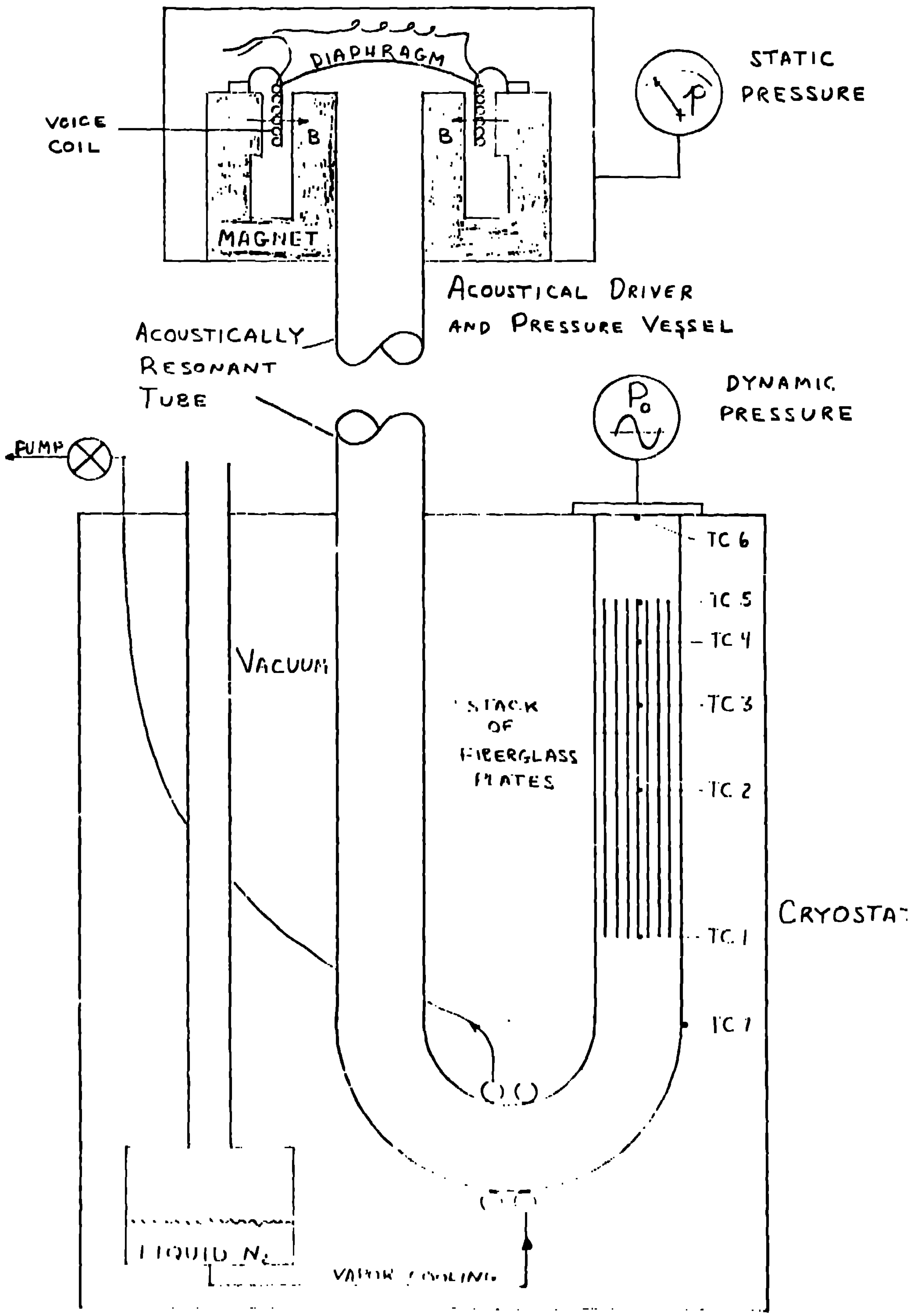
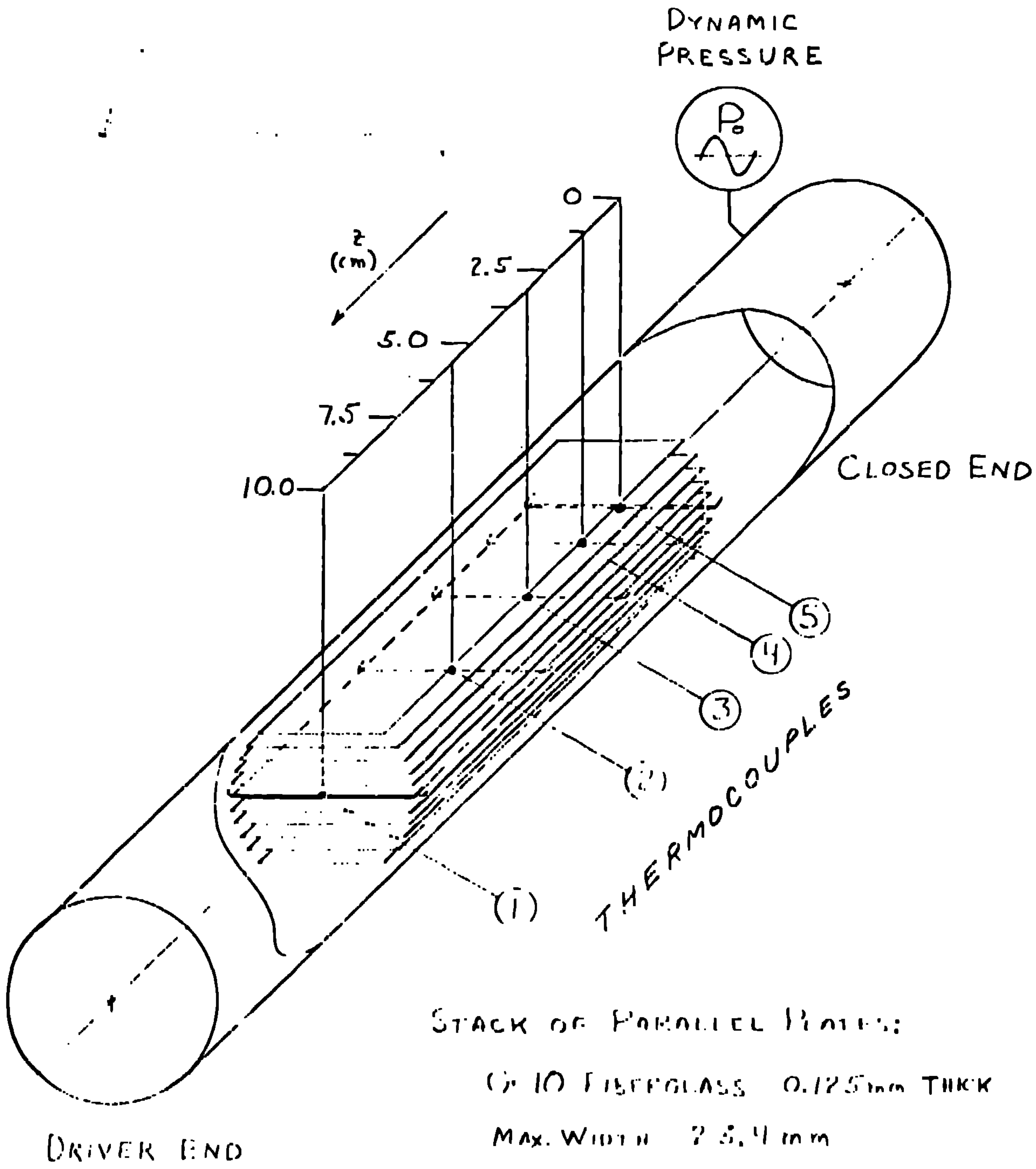


FIG 1



STACK OF PARALLEL PLATES:

G-10 FIBERGLASS 0.125 mm THICK

MAX. WIDTH 25.4 mm

PLATE SEPARATION 0.91 mm

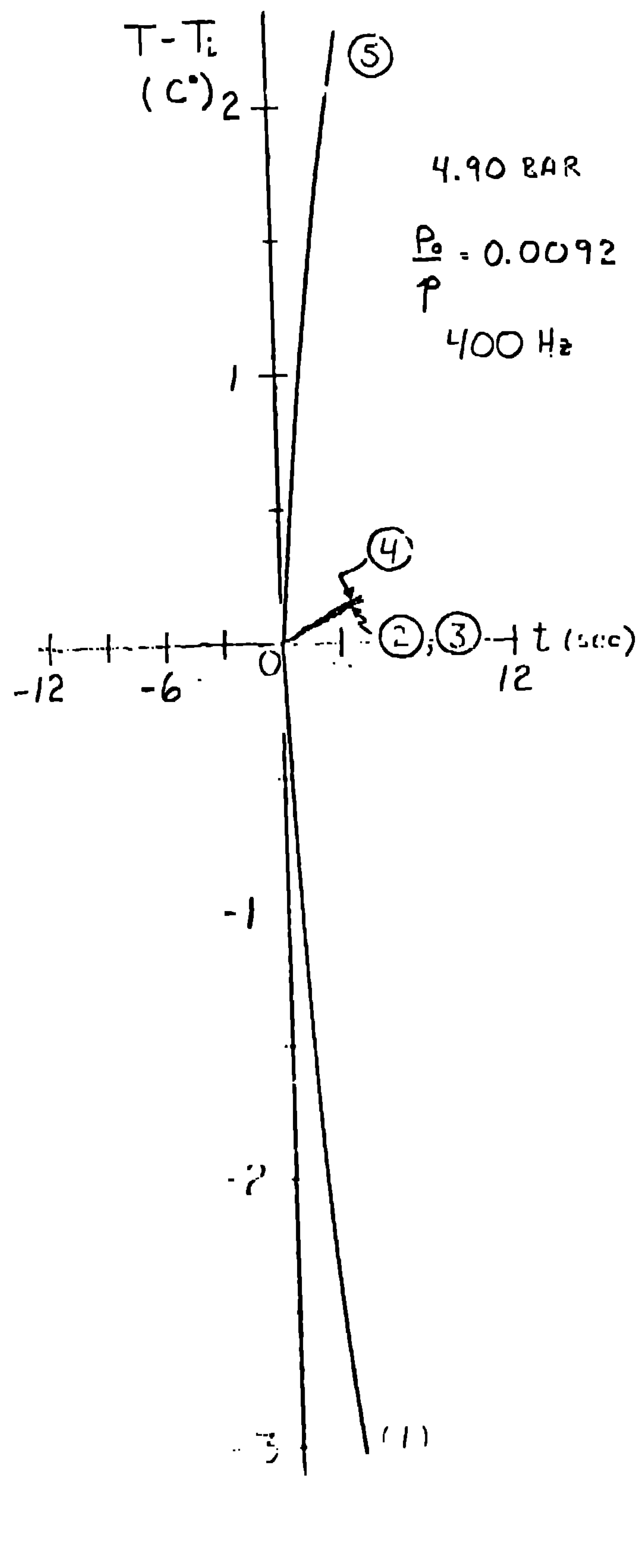
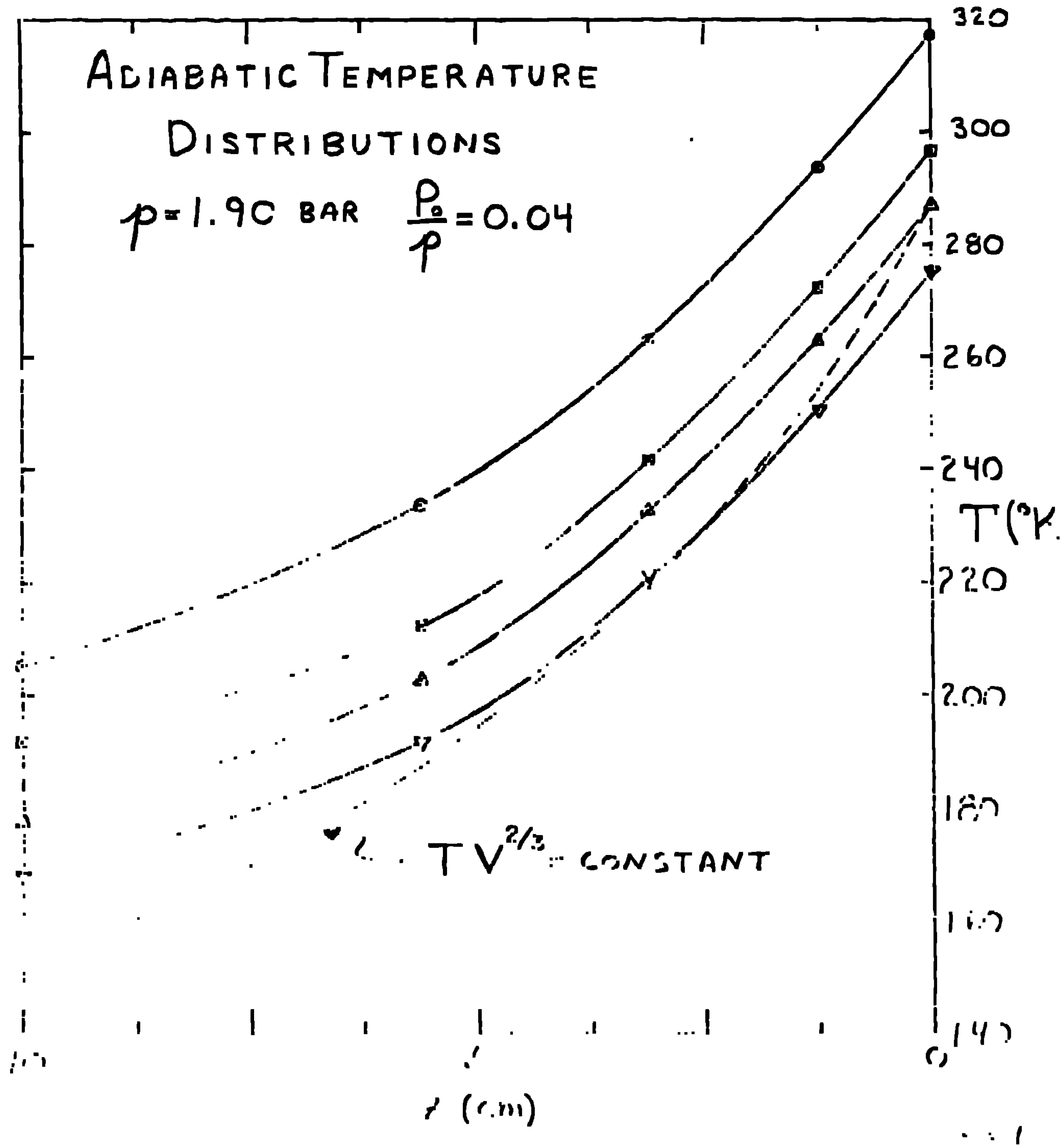


Fig 3



ADIABATIC TEMPERATURE DISTRIBUTIONS
p = 1.90 BAR P₀/p = 0.04
TV^2/3 = CONSTANT
T(°K)
320
300
280
260
240
220
200
180

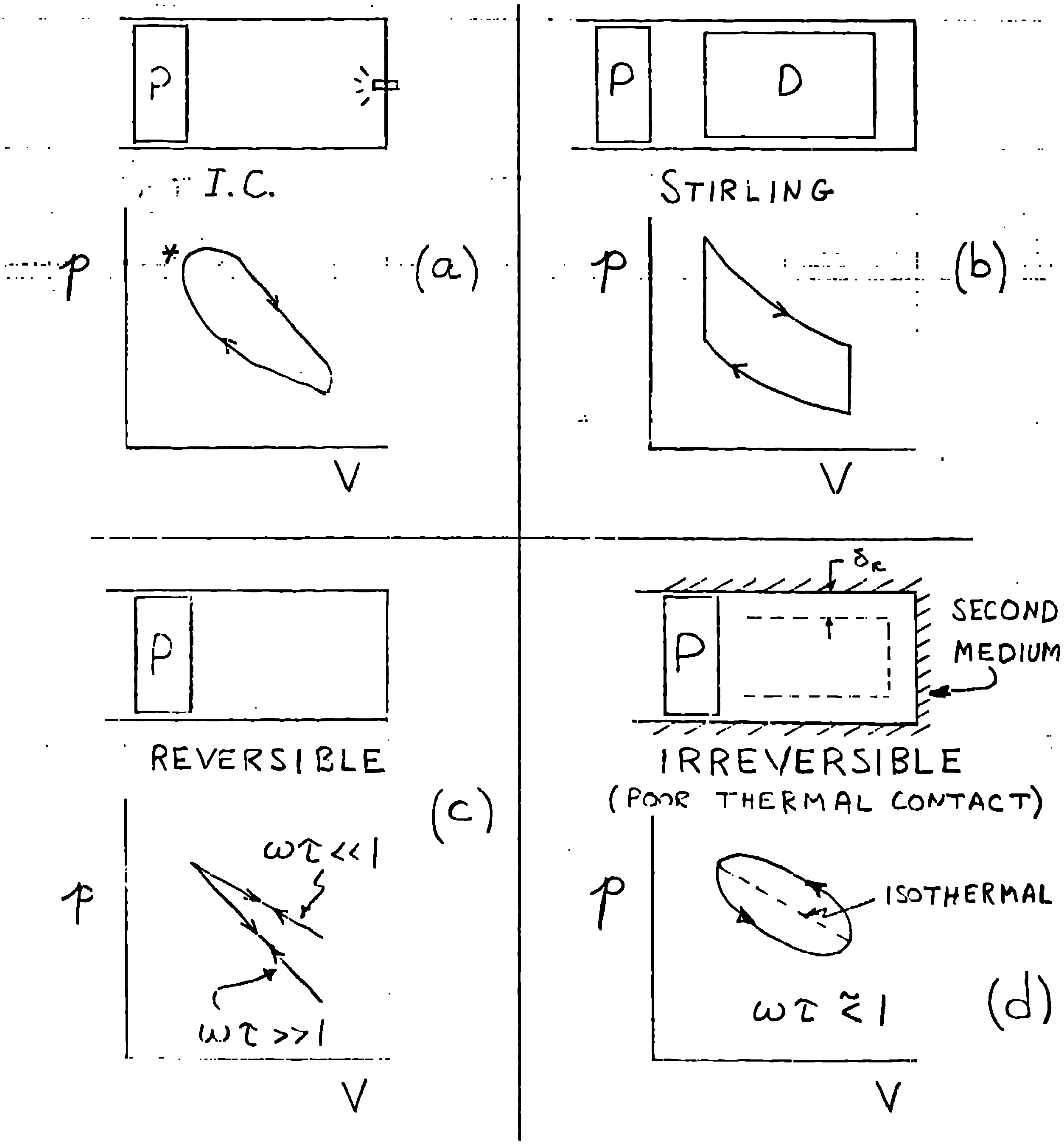

P
I.C.
p
V
(a)
P
D
STIRLING
p
V
(b)
P
REVERSIBLE
(c)
p
$\omega\tau \ll 1$
$\omega\tau \gg 1$
V
δ_κ
P
SECOND MEDIUM
IRREVERSIBLE
(POOR THERMAL CONTACT)
p
ISOTHERMAL
$\omega\tau \gtrsim 1$
(d)
V

Fig.5

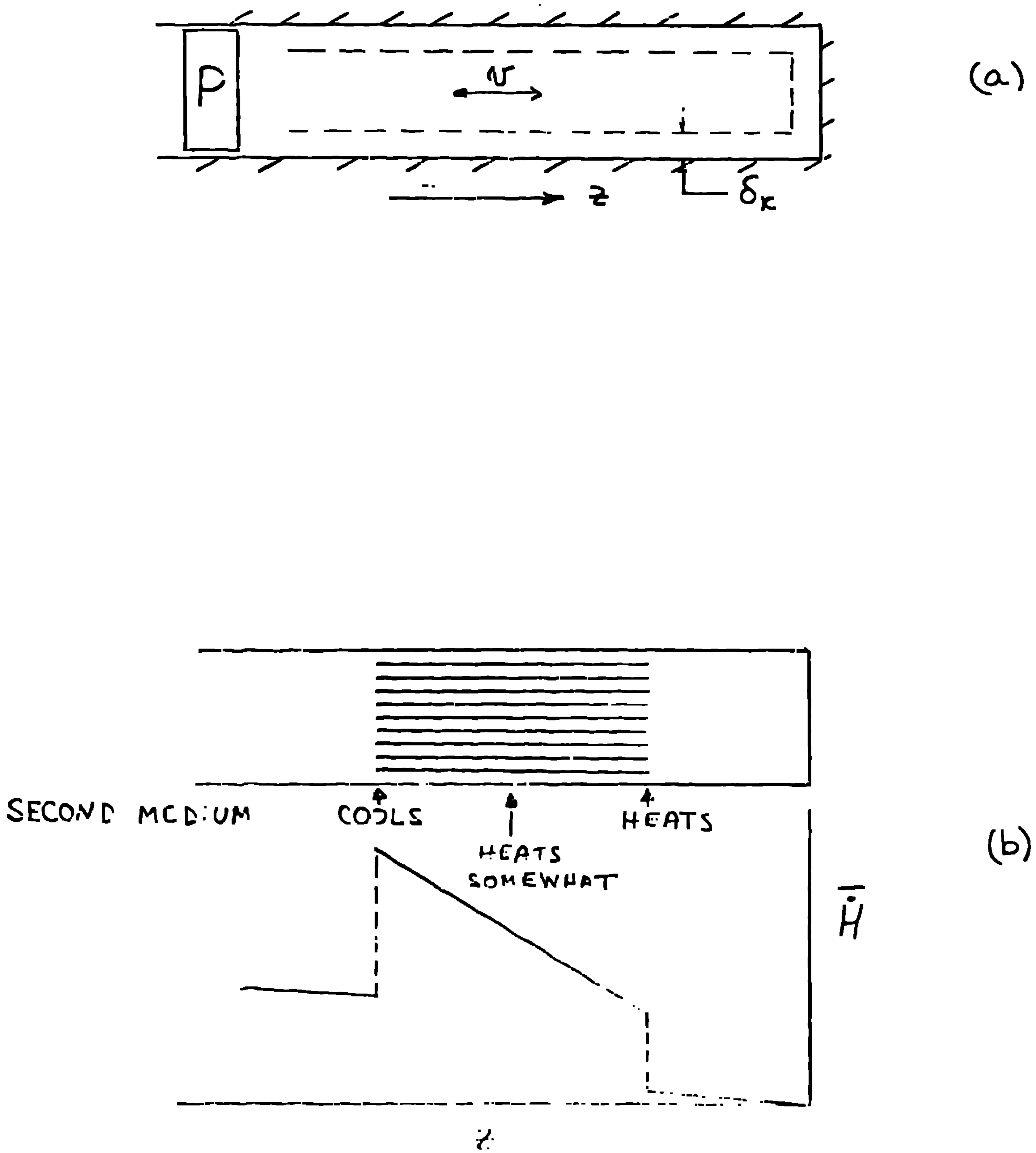


Fig 6

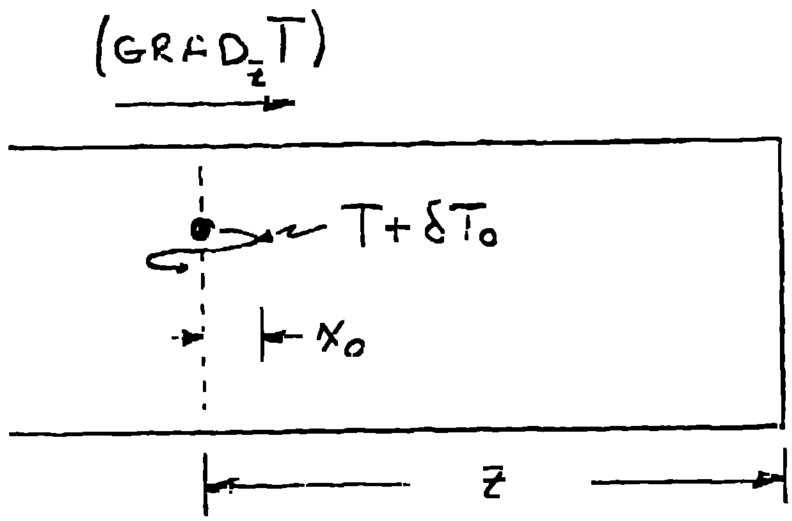


FIG. 7